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The Condition Monitoring of an Upstream Oil and Gas Dry Screw Compressor Drive Train and its Impact on the Control System

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ABSTRACT

Variable speed oil free screw compressors can be used in upstream oil and gas applications such as gas collection and flare gas recovery. The critical nature of these applications is such that the owners closely monitor the drive train of the package as long un-planned stoppages are extremely expensive. If any of the monitored parameters start to deviate from the norm then shut-downs and services can be scheduled in a timely manner preventing costly stoppages.

This paper first describes the parameters monitored on the drive train as required by the international standards used by the oil and gas companies. The paper then leads onto an in-depth description of the instrumentation mounted on and even within the drive train, incorporating the main drive motor, gearbox and compressor, and how the instrumentation is monitored by a package machine condition monitoring system that interfaces with the package control system. As the final location of the instruments is often in extremely hazardous environments such as highly corrosive offshore marine environments or abrasive hot desert environments, and the compressors themselves are often compressing highly combustible gas such as methane or even hydrogen, the protection methods employed by the instruments to prevent explosion is then discussed.

The multiple communications networks that allow the condition monitoring system to send its values to the package control system and the client's distributed control system are also discussed. This discussion includes how the measured values make their way from the stand alone machine condition monitoring system to the compressor control system and how the two systems integrate with each other to protect the package before annunciating the system status to the client's distributed control system.

Finally a full description is given on how these measured values impact on the running of the package with respect to the compressors service life and the control systems ability to maintain the process setpoints. A solution to the presented problem is then offered.

1. INTRODUCTION

The screw compressor is a positive displacement rotary machine which is made of three main parts: the compressor casing and a pair of meshing helical lobed rotors all of which are fitted tightly together. The clearance between the rotors themselves and the clearance between the rotors and the compressor casing is very small to reduce internal leakages which affect efficiency. The package main drive motor, in conjunction with a gearbox, drives the compressor's internal timing gears which turn the rotors. The rotors are designed in such a manner that the cavity formed between the rotors is continually reduced during the revolution. Obviously any gas trapped within the cavity would be compressed as the rotors revolved. The process gas enters the compressor through a large inlet port, into a large cavity between the rotors and then as the rotors spin the cavity moves away from the inlet port and reduces in size where at the end of the rotor the cavity is at its smallest and passes over a discharge port to allow the

compressed gas to escape. The compressor produces a continuous flow of pulses of high pressure each time a high pressure cavity of gas passes over the discharge port.

In an oil free machine there is no lubrication oil injected into the working chamber to lubricate the rotor motion; instead the rotors are kept out of contact with each other by means of timing gears in a separate chamber. The compressor cooling jacket, bearings and timing gears are all still lubricated but kept isolated from the working chamber. To isolate the working chamber from the bearings a series of mechanical seals are introduced between them. The cooling of an oil free compressor is not as efficient as the cooling of an oil injected compressor, even with a cooling jacket and hollow rotors to aid cooling. This lubrication oil supply is controlled by the package control system in general accordance with API 614 chapter 6 (2008). Oil free screw compressors, while more expensive than oil injected machines, they find their use in processes where contamination of the process gas is to be avoided.

Screw compressors can be used in upstream, mid-stream and downstream oil and gas applications. The critical nature of these machines, both in cost to the end user in the event of an unplanned shutdown and in terms of the explosive nature of the process gas, means that condition monitoring equipment is used to constantly monitor the health of the drive train.

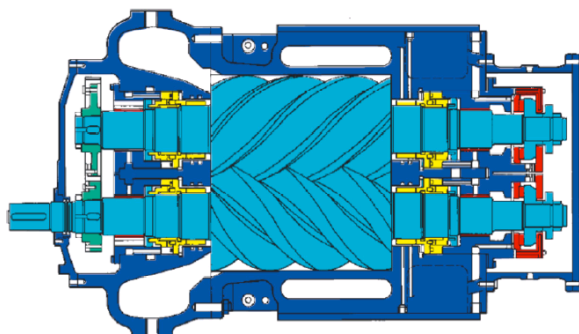


Figure 1: A cut away diagram of an oil free screw compressor

2. Machine Condition Monitoring Instrumentation

2.1 Main Drive Motor

On the main drive motor (MDM) the windings are protected by resistance temperature detectors (RTDs) located within the windings themselves. Duplex (two: one active and one spare) RTDs are provided for each winding in the event that if the primary RTD fails the cabling can simply be switched to the spare. The RTDs are routed from the windings housed within the motor stator out to their own junction box separate from the main terminal box.

Within the RTD junction box are terminals to allow easy field connection. The junction box is certified as part of the motor certification for use within a zone 2 and zone 1 environment as designated by BS EN 60079-10 (2007). From the RTD junction box a blue sheathed cable is run directly back to the unit control panel (UCP) carrying the three RTD signals. The cable has a blue sheath as it contains intrinsically safe signals and all field cabling is terminated using flameproof nickel-plated brass armour glands complete with PVC shroud to protect it from the surrounding environment.

The signal is then passed through a galvanic isolator, which limits the amount of energy supplied to the hazardous area, before being terminated into the MCM rack. In order to conform to the BS EN 60079-1(2007) certificate supplied with the motor [more specifically the certificates schedule of special conditions of use] then the motor must have temperature protection operational in order to trip the machine in the event of an over temperature situation. The motor is operated in environments where explosive gases could be present which have low ignition temperatures which is one more reason, other than to protect the motor windings from damage, to protect the motor from overheating. There are various methods to achieve this protection such as the RTDs could have been terminated into the VSD to protect the motor but to keep all drive train temperature monitoring following the same convention they were terminated into the MCM rack.

A Machine Condition Monitoring (MCM) piping and instrumentation diagram (P&ID) is shown in Figure 2. All MCM instruments are protected from explosion by using intrinsically safe methods (BS EN 60079-11, 2007) and as a result the instruments could be deployed into an extremely hazardous gaszone 0 environment (BS EN 60079-0, 2006).

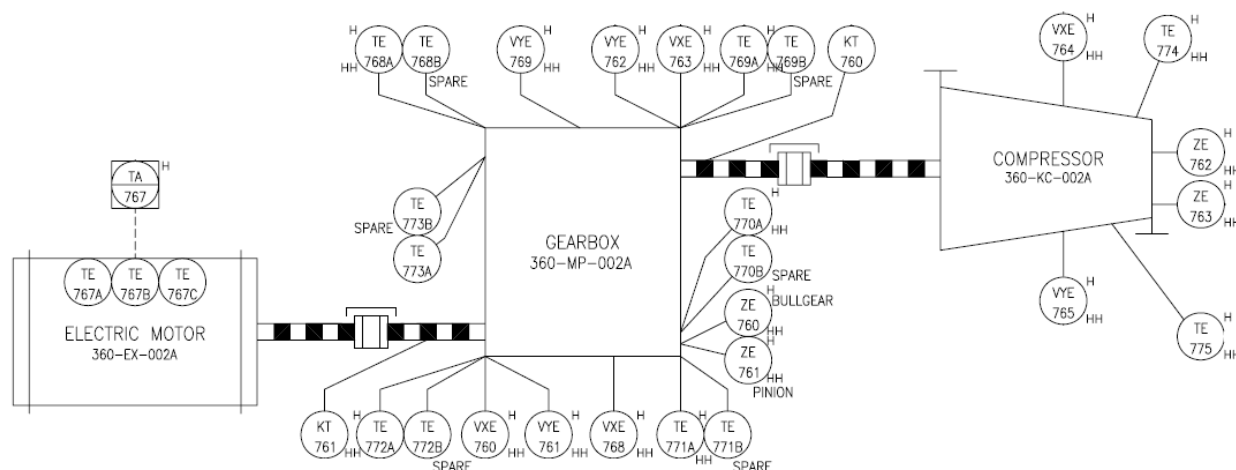


Figure 2: Piping and Instrumentation Diagram (P&ID) of the MCM instruments used on the entire drive train

2.2 Gearbox

The gearbox is similarly fitted with duplex RTDs on the pinion Non Drive End (NDE) bearing; pinion Drive End (DE) bearing; pinion thrust bearing; bullgear NDE bearing and finally bullgear DE bearing. These RTDs monitor the temperature of the gearbox bearings to ensure that the bearings are in good condition as worn bearings or bearings starved of lubrication oil would result in a high bearing temperature due to an increase in friction. The gearbox RTDs are wired back from their bearings back to a junction box mounted on the side of the gearbox. The junction box is certified on its own to be used within a zone 2 or zone 1 environment but as it only contains intrinsically safe signals it can be deployed within a zone 0 environment. The gearbox MCM instruments are not limited to temperature monitoring; vibration monitoring is also provided in accordance with API 670.

2.3 Compressor

The compressor MCM instrumentation includes male and female rotor thrust bearing temperature RTDs run from the bearing and terminated into small Ex e junction boxes (BS EN 60079-7, 2007). From here the signals are taken to the main MCM junction box. The RTDs range is from 0 – 200°C. This range allows complete monitoring of the lubrication oil temperature of the bearings and within the range is a high temperature alarm limit and a high-high temperature trip limit that will first alert the operators to an abnormal condition then trip the machine before damage can occur to the bearings.

The casing accelerometers, both the horizontal and vertical axis accelerometers, are fitted to the compressor casing and a bespoke, spiral wound armour cable is run from the accelerometer housing back to the junction box. The accelerometers require a -24 Vdc supply and send a scaled signal back to the MCM rack in the order of 10.2 mV/(m/s²). The range of the accelerometers is from 0 – 490 m/s² peak. Within this range there is a high vibration alarm limit and high-high vibration trip limit that will first alert the operators to an abnormal condition then trip the machine before damage can occur.

The female and male rotor axial displacement probes use a proximity transducer system to transmit the measurement of the rotor axial displacement back to the junction box. The system itself consists of a probe, extension cable and proximator sensor. The system provides an output voltage directly proportional to the distance between the probe tip and the observed conductive surface. It is capable of both static and dynamic measurements, and is primarily used for vibration and position measurement applications on fluid-film bearing machines but can also be used for keyphasor applications such as on the low speed and high speed shaft on the gearbox.

As with the other MCM signals a blue sheathed cable is run directly back from the junction box to the UCP carrying the signals. The signals are then passed through galvanic isolators before being terminated into the MCM rack. The compressor with its MCM instrumentation fitted is shown in Figure 3.

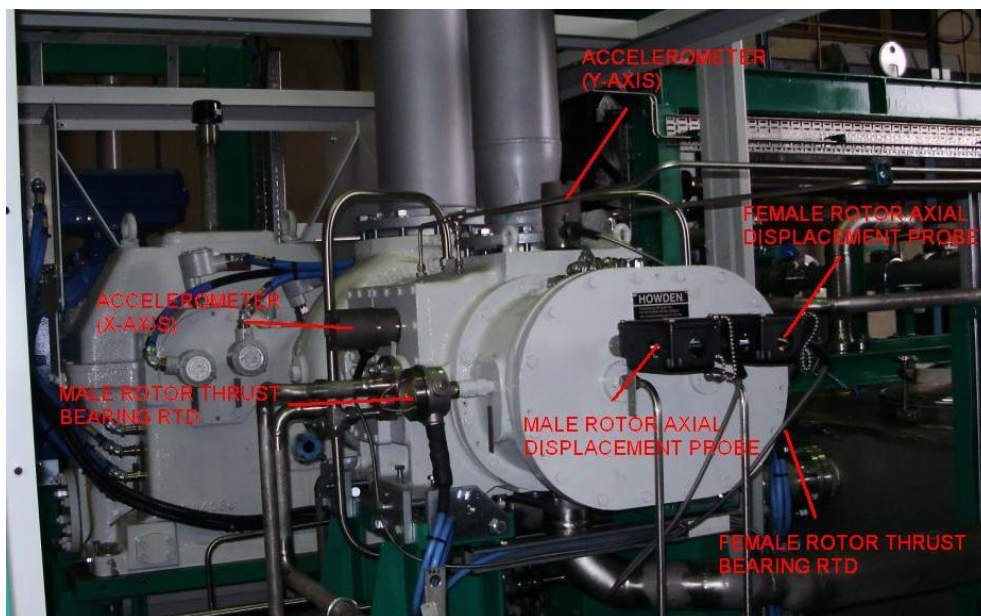


Figure 3: Layout of machine condition monitor equipment on the compressor. Shown are the accelerometers, bearing RTDs and rotor axial displacement probes

2.4 UCP Rack

All of the MCM system instruments are cabled back to the Bentley Nevada 3500 MCM rack. This rack consists of a 15 slot chassis mounted within the base frame of the UCP. The monitoring modules are inserted from the front but all cable terminations are to the rear of the modules. The MCM rack is powered using dual redundant power supply units (PSU) so that in the event one fails, the other immediately assumes duty so that the MCM system does not trip. The power supply modules can be removed and replaced under power so long as the redundant unit is fitted and powered.

The MCM rack has a rack interface module (RIM) for programming and configuring the monitoring modules, relay modules and communication modules. The module takes around twenty seconds to reset the system which can become frustrating when multiple resets are required during the configuration period. After connection to the RIM via a laptop with the configuration software installed the ability to bypass alarms and trips of the individual inputs is available. This gives the ability to commission the machine, set-up the rotor axial displacement for the first time for instance, without being hampered by nuisance alarms. This is a very powerful feature and actual alarms or trips that would normally shutdown the machine could be suppressed using this feature and ruin the machine. The RIM also has a trip multiply input that if switched, and if configured in the software, multiplies the alarm and trip settings by either two or three. This trip multiplier can be used during the start-up of the main drive motor to allow the machine to move through its natural frequencies if required. The RIM has a status output contact that is wired back to the PLC that alerts the package control system if the MCM system fails. To prevent tampering the RIM module has a key switch that once the module has been commissioned the key switch can be locked and the key removed to stop access. Two communications ports are available on the RIM, one serial port is situated on the front of the module which allows system configuration and a second Ethernet port is located on the rear of the module to allow communication between the system and a Bently Nevada "System One" site monitoring package.

The temperature monitor modules are where all RTDs of the MCM system are terminated. The radial vibration, axial displacement and casing accelerometer instruments are wired back to the proximator seismic modules. The sixteen channel relay output module is installed to send hardwired signals to the PLC in the event of a serial communication failure. A communications module is fitted to provide a Modbus serial link to the PLC which allows individual instrument values and statuses to be transmitted over it. The physical layer of the system consists of one

master and one or several slave nodes “daisy chained” together. On this simple point to point network, commonly used in automation systems, the MCM rack is the slave node and the PLC is the master and is a very common topology (Kang and Robles, 2009). The Modbus link passes the measured values into a register within the PLC.

3. Data Acquisition and Analysis

Trends of the values measured by the MCM instrumentation are used to investigate the behavior and response of the machine to various process conditions. Trends of the package machine condition monitoring instrumentation during various stages of the project were gathered from the initial run up to and including commissioned runs on process gas in the Sahara desert.

The previous section discussed getting data from the package instrumentation, either from the MCM instrumentation or the process instrumentation and putting it into registers within the PLC. When a laptop is connected to the PLC, done using the ControlNet to Ethernet bridge on the HMI, the laptop can be used to ‘go-online’ to the PLC. This allows the engineer to view the PLC programme live and watch values change as they happen and logic bits energise and de-energise. It also allows the engineer to create trends of the desired process parameters.

The selected process values are recorded by configuring pens on the trending screen. The pens are then pointed to a particular tag within the program, the tag being the alias given to the MCM values to be recorded. The trends are a powerful tool in the analysis of plant performance. Clients contact the after sales service department describing a particular problem and send through a DCS trend as evidence. DCS trends invariably have long sampling times with anywhere from twenty seconds to five minutes per sample or sometimes even longer. This is due to the DCS having to record an entire refinery, or platform, process values and a high sampling rate would rapidly fill its memory. So in order to get a true picture of what is happening within the package an engineer is deployed to ‘go-online’ with the package PLC and record the actual process problem.

3.1 Speed

The PLC outputs an analogue 4-20 mA signal to the ABB ACS 800 variable speed drive (VSD). This VSD then increases the motor speed by increasing the frequency of the alternating current. This in-turn increases the speed of the compressor which is measured by the gearbox high speed shaft. The keyphasor sends the speed measurement back to the Bently Nevada MCM system and once registered here the circle is complete by sending the measured speed back to the PLC via the Modbus serial link to be trended. There is a small lag between sending the speed reference signal to the VSD and the main drive motor actually achieving this speed due to both the PLC and VSD controller settings and the inertia of the motor’s rotor, the gearbox gears and shafts and finally the oil free screw compressors timing gears and rotors. The measurement of the increased speed and the registering back to the PLC via the MCM system incorporates a further small delay.

3.2 Rotor Axial Displacement

During the completion of the bareshaft machine test the compressor has not been furnished with all of the machine condition monitoring instrumentation. During this test the axial displacement probe mounting holes are plugged so no information can be acquired on the rotor displacement at this stage.

In the run up to the package test when a series of test runs are being performed, and the plugs have been removed and the axial displacement probes have been fitted, the probes can then be set-up by the test engineer. To set-up these probes, both the male and the female probes, it must first be understood that there is a certain amount of float on the compressor rotors and that the rotor position at start up is not necessarily the nominal rotor position during normal operation due to things like process loading and thermal expansion. For this reason, the compressor is allowed to start without any alarms and trips activated regarding the rotor axial displacement. After running for around half an hour, the system is deemed to have warmed up and the rotor should have floated to its nominal position so set-up can commence.

3.3 Temperature

The compressor journal bearing temperatures are monitored during the initial runs as a faulty bearing will first be highlighted by a high bearing temperature. During the package prestart operation, as the lubrication oil is being

pumped around the system without any heat being generated at the bearing, the bearing temperatures should match the manifold lubrication oil temperature.

Once the package has started and the compressor rotors are spinning at over 8000 rpm the bearings generate their own heat and this, coupled with the lubrication oil being warmed as it flows over the compressor cooling jacket, leads the bearing temperature to rise. Conversely, when the compressor is stopped either due to a normal stop or a process trip stop, the bearings rapidly cool back down to the manifold temperature as they are fed continuously with lubrication oil during these sequences. However, it must be noted that if the compressor package is ever emergency stopped the auxiliary motors on the package also cease which prevents lubrication oil being fed into the bearings which would see the bearing temperature rise.

3.4 Casing Vibration

The compressor casing vibration is monitored by using the accelerometers mounted on the side and top of the compressor casing. These values are routed back to the PLC in the same manner as the previously discussed machine condition monitoring instrumentation.

It was important to understand not only the vibrations inflicted on the compressor during start up to its minimum speed, but also as it ramped from minimum speed up to full speed. It is important to remember that during the compressor start sequence the package runs up to 50 % as quickly as possible with the machine totally unloaded, then after the operator has dealt with the manual valves within the process, the package is allowed to automatically load and vary its speed.

In order to understand the relationship between the compressor speed and vibration, trends were recorded of the package during its initial 0 – 50 % ramp and of a ramp between 50 – 100 %. The 4-20 mA speed reference signal sent to the VSD from the PLC was manually inputted to 50 % or 12 mA for the purposes of this part of the test and stays at this value throughout the trend. This means that as soon as the VSD is given a start pulse from the PLC it will drive the compressor to this speed as quickly as possible.

When the start pulse energises the VSD, the gearbox high speed shaft values pickup quickly and reaches 50 % speed in marginally over ten seconds. During this time the casing accelerometers experience no excessive levels of casing vibration the vibration in the horizontal plane is considerably greater than the vibration experienced in the vertical plane.

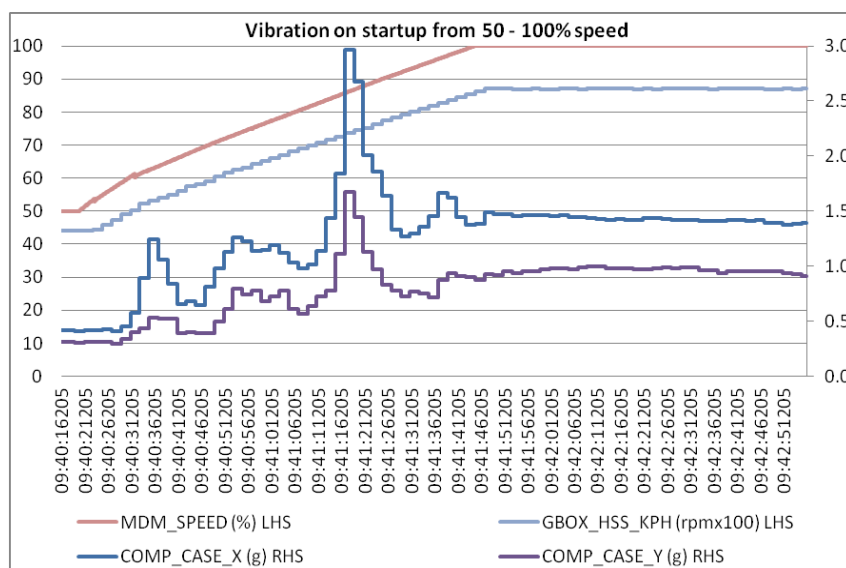


Figure 4: The vibration versus speed values recorded at the compressor when starting at 50 and ramping to 100% of maximum speed during the factory test

Unfortunately this was not the case during the recorded ramp between 50 % and 100 % as can be seen from Figure 4. The system was in a state of equilibrium with the speed reference signal showing 50 %, the gearbox high speed

shaft showing approximately the corresponding speed of 4411 rpm, and the compressor X and Y casing accelerometers measuring 0.45 g and 0.3 g respectively.

The 4-20 mA speed reference signal was then removed from manual control where the value is imputed via the HMI and put into automatic control which had been setup to temporarily use a 4-20 mA signal injected from a milli-amp current source as the speed reference signal. The 12 mA speed reference signal was then slowly increased up to 20 mA with a corresponding rise in the gearbox high speed shaft keyphasor measurement up to 8715 rpm. However, during this particular ramp the trend demonstrates a massive spike between 7160 and 7828 rpm in terms of rotor shaft speed, or 82.2 % and 89.8 % in terms of PLC PID block output, or lastly 41.1 Hz and 44.9 Hz in terms of VSD frequency output. The vibration peak in the 'x' or horizontal direction during this speed is 200 % the nominal vibration when the motor is running at full speed and while the 'y' or vertical vibration peak is only around 170 % of the full speed nominal vibration it is still far from satisfactory.

It can also be seen from Figure 4 that three smaller rises in vibration level occur during the ramp with peaks around 60 %, 76 % and 95 % however these peaks either do not exceed, or only marginally exceed the nominal vibration at full speed and as a result of this are excluded from the analysis.

Firstly there is a need to account for the increased level of vibration in the horizontal direction or conversely the reduced level of vibration in the vertical direction. The compressor is fixed to a plinth using vertical fixings with less support in the horizontal plane meaning the compressor is more stable in a vertical axis than it is in the horizontal axis.

In API 619 (2004) a note is offered to the reader that the machines structural support may adversely affect the rotor vibration amplitude. While this seems to be the case regarding the compressor stool the document was intending to highlight that as a result of this the vibration measured during machine tests and even package tests on the factory floor must be compared with results taken from the packages final location as each may be influenced by the surrounding support structure or lack thereof.

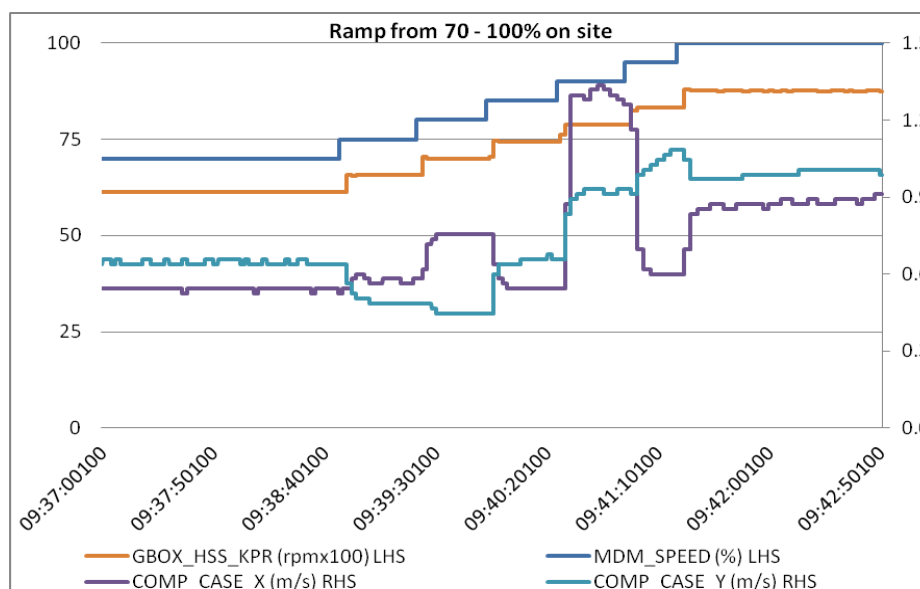


Figure 5: The vibration versus speed values recorded at the compressor when starting at 50 and ramping to 100% of maximum speed during the factory test

Before discussing Figure 5 it should be noted that the vibration units have now been changed to their integrated speed values (m/s) instead of the (g) measured during the factory tests. Again from the ramp taken at site as the package is ramped from 70 % - 100 %. This time no current milli-amp injector was available and instead the speed control PID block was put into manual and the speed reference values were manually inputted to the HMI. The

speed reference signal was then slowly increased in increments of 5 % up to 100 % with a corresponding rise in the gearbox high speed shaft keyphasor measurement up to 8757 rpm.

The reader will also note that prior to starting the ramp the vertical plane vibrations are larger than the horizontal plane vibrations which is the opposite of the vibration levels witnessed in the factory. This change can be attributed to the fact that the package is now bolted to a concrete plinth reducing any lateral movement but also to the fact that the client has now attached substantial amounts of process piping to the package suction and discharge piping that alters the dynamics of the package pipework which is connected vertically to the compressor.

During this particular ramp the casing horizontal vibration raises to a small peak around the 80% of full speed but like the smaller peaks discovered during the factory ramp test this can be discarded from the analysis as it remains smaller than the full speed vibrations levels. However, again the ramp displays a large spike in horizontal vibration readings but this time between 7894 and 8252 rpm in terms of rotor shaft speed, or 90.1 % and 97.3 % in terms of PLC PID block output, or lastly 45.1 Hz and 48.7 Hz in terms of VSD frequency output.

The vibration peak in the 'x' or horizontal direction during this ramp has decreased significantly but is still nearly one and a half times the nominal vibration when the motor is running at full speed and with the 'y' or vertical vibration peak also reduced so it is now only around 110 % of the full speed nominal vibration at its peak. It should be noted that there were no excessive vibrations recorded either from the horizontal or vertical casing accelerometers during the stopping sequence.

4. Solution and Conclusion

Only after developing the control system and in particular its ability to acquire and record the process and machine condition monitoring values could a practical investigation for critical speeds within the operating range be conducted. Evidence of the critical speed was found while testing on the factory floor but it was not until the screw compressor package was sited, levelled and grouted to its plinth in the Sahara desert with all of its process pipework connected until the true extent of the critical speed within the operating range could be gauged.

From analysis of the factory ramp and the site ramp the vibration levels appears to cover a 7 % window of the speed range but has shifted up the speed range from the factory to the site and while the overall level has reduced it will still be prudent to remove this speed window from the control systems operating range.

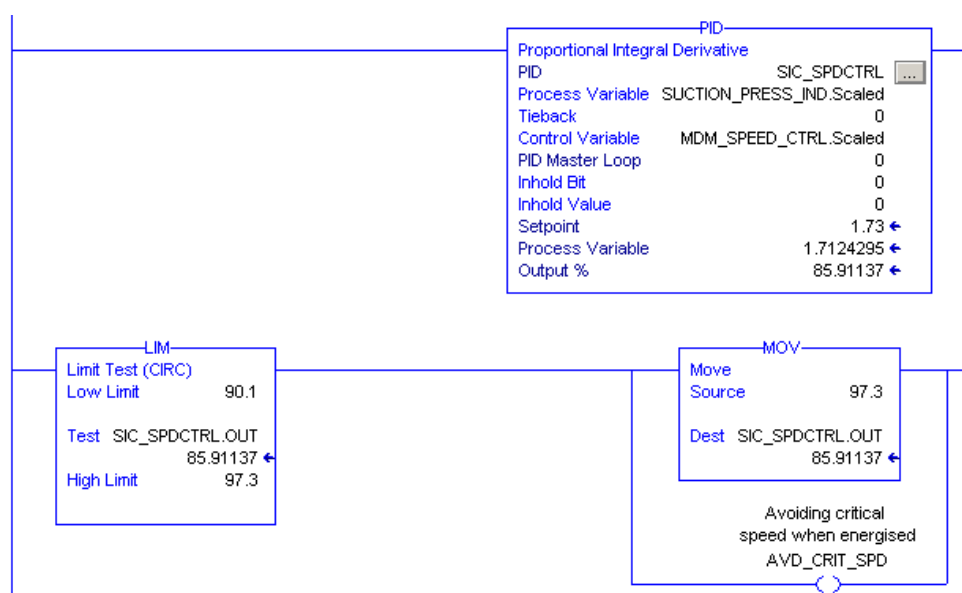


Figure 6: Using the PLC ladder logic to step the VSD over the critical speed found during the site test run

By removing this speed window from the compressors operating range its ability to control the client's suction pressure at the required setpoint is now limited. Avoiding the critical speed between 45.1 and 48.7 Hz can be achieved by utilising one of two methods. One method is by using the PLC programming software and modifying the code to step over this critical speed. As mentioned previously the VSD had its own PID controller within the PLC code. This PID controller outputted to a register, as a percentage, how fast the VSD was to run before an analogue scaling block changed the percentage to a 4-20 mA signal that was sent to the VSD. Unfortunately the PID block within the PLC programme does not have the ability to configure output values that are to be avoided. So the code was developed in the next rung below the speed control PID to monitor the output from the speed controller and if the output is between 90.1 and 97.3 %, which corresponds to the 45.1 and 48.7 Hz, then a value of 97.3 % is moved into the PID output register instead. On the same rung as the move instruction is moving the value of 97.3 % into the PID controller output a Boolean logic bit with the tag AVD_CRIT_SPD is energised showing that the PLC is now having to avoid the critical speed. This has the effect where the PLC never instructs the VSD to run at or around the critical speed but instead steps beyond it to run slightly faster. Figure 6, shows the PLC code used to avoid the critical speed.

Stepping the equipment around the critical speeds and thus avoiding peaks in vibration is a common technique. Indeed the other method that could have been used to avoid the critical speed was the ABB variable speed drive's built-in critical speed avoidance software function (ABB, 2009). This function is not uncommon with most VSD manufacturers offering something similar. However, in this application by simply avoiding the critical speed, including by using the method adopted by the authors by means of the PLC code, it has a negative effect on the suction pressure the package is required to maintain.

By running the machine faster than the desired speed the suction pressure drops below the suction pressure setpoint and hence eventually slows the machine down to let the suction pressure build back up. When the suction pressure set-point is achieved and the machine again needs to run at the critical speed the control system again increases the reference speed to avoid the critical speed and so the cycle continues with the VSD hunting around the critical speed continually slowing down to let the suction pressure build back up then running too fast and thus reducing the suction pressure. If a value of 90.1 % was used i.e. the slower of the two threshold speeds then the suction pressure would have artificially built up as it ran below the desired output. Eventually the measured suction pressure would cause the controller output to breach 97.3 % with the VSD then increasing speed to dissipate the suction pressure. As the VSD speed increased and the suction pressure duly dropped then the controller output would again fall back into the critical speed band forcing the output back to 90.1 % and the hunting cycle would start again.

A solution to prevent this hunting and allow the motor to run constantly at a higher speed yet maintain the desired suction pressure was found. The control philosophy stated earlier that when the VSD is at minimum speed and the measured suction pressure is still lower than the suction pressure set-point then the package recycle control valve is opened. This was to allow the suction pressure to build back up by recycling the discharge gas back to the suction side. The recycle control valve is again controlled by its own PID controller within the Allen-Bradley Control Logix PLC. An analogue 4-20 mA signal is sent from the control panel to the Ex ia rated current to pneumatic converter on the control valve as a reference signal where 4 mA represents the control valve being 0 % open and 20 mA represents the control valve being 100 % open. The PLC programme was changed so that when the speed controller output is between 90.1 and 97.3 %, then not only is the speed reference signal set at 97.3 % but the recycle control valve is now also enabled to control. This allows the compressor to run faster, adversely affecting the suction pressure, but then compensating this at the same time by opening the recycle control valve to allow some of the additional discharge gas back to replenish the gas from the suction side that is removed due to the faster VSD speed. Again, looking at if a value of 90.1 % was used as the fixed speed and the suction pressure was allowed to build up there would have been no way of compensating for this with the recycle control valve or indeed any of the existing plant or instrumentation other than to allow the compressor to increase speed once the high end of the critical speed had been breached which simply results in the VSD hunting around.

When the recycle control valve is in operation to replenish the excessively depleted suction pressure and compensate for the increased speed but the speed controller moves the PID controller output away from the critical speed, which could be as a result of change in suction pressure set-point, change in suction pressure from the well-head or upstream disturbances such as flare control valves then the system responds accordingly. The VSD will change speed and again this will depend on the type of disturbance where an increase in the suction pressure set-point would see the VSD slow down and a decrease in the suction pressure set-point would see the VSD speed up. At the same

time as the VSD is responding to the system disturbance the recycle control valve will close as the recycle control valve is only available to operate when the speed controller output is between 90.1 and 97.3 % or equal to 50 % which equates to the VSD minimum speed. At all other times the PLC sends a 4 mA signal to the control valve to ensure it stays closed.

Normal control of the recycle valve is enabled when the VSD equals 50 % speed, then the recycle valve will be used to try and control the suction pressure in a cascade style control technique. However, as just discussed, when the speed controller output is outputting a signal between 90.1 and 97.3 Hz the recycle valve control must be enabled to replenish the suction pressure so the AVD_CRIT_SPD bit is then used to energise the recycle valve controller. During normal operation when the suction pressure is being controlled by means of the variable speed drive a value is written to the recycle valve output that prevents it from opening. Obviously, when the speed controller is outputting a value between 90.1 and 97.3 % and the system is trying to open the valve, this value has to be prevented from getting to the valve output so an AVD_CRIT_SPD normally closed logic bit is inserted into the rung between the comparison and move instructions that write this value. When the AVD_CRIT_SPD output is energised in the speed controller routine, this normally closed logic bit opens and prevents the valve from being instructed to close, allowing the recycle valve controller to operate freely.

Failure to avoid the critical speed would mean the compressor would suffer the increased vibration levels and the resultant consequences. The time spent running at the higher vibration levels would have depended on a combination of factors including: the well-head suction pressure, system disturbances and suction pressure set-point. Running a compressor at high vibration levels reduces the life of the bearing due to the increased forces which is why expensive machine condition monitoring systems are installed and oil analysis is carried out on the lubrication oil.

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